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BEARING TESTER DATA COMPILATION  
ANALYSIS, AND REPORTING AND  
BEARING MATH MODELING

MONTHLY PROGRESS REPORT  
FOR  
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## FOREWORD

This quarterly progress report was prepared by SRS Technologies under Contract No. NAS8-36183 for the George C. Marshall Space Flight Center of the National Aeronautics and Space Administration. The work was administered under the technical direction of the Materials and Processes Laboratory, Engineering Physics Division with Mr. Fred J. Dolan acting as project manager.

This report describes the work performed by SRS Technologies during the July, 1986 reporting period. Mr. Joe C. Cody served as the SRS Technologies Principal Investigator. The SRS project personnel who made major contributions to this report include:

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## 1.0 SUMMARY

Integration of heat transfer coefficients, modified to account for local vapor quality, into the 45 mm bearing model has been completed. The model has been evaluated with two flow rates and subcooled and saturated coolant. The evaluation showed that by increasing the flow from 3.6 to 7.0 lbs/sec the average ball temperature was decreased by 102°F, using a coolant temperature of -230°F. The average ball temperature was decreased by 63°F by decreasing the inlet coolant temperature from saturated to -230°F at a flow rate of 7.0 lbs/sec. Since other factors such as friction, cage heating, etc., affect bearing temperatures, the above bearing temperature effects should be considered as trends and not absolute values.

The two phase heat transfer modification has been installed in the 57 mm bearing model and the effects on bearing temperatures have been evaluated. The average ball temperature was decreased by 60°F by increasing the flow rate from 4.6 to 9.0 lbs/sec for the subcooled case. By decreasing the inlet coolant temperature from saturation to -245°F, the average ball temperature was decreased 57°F for a flow rate of 9.0 lbs/sec. Thus, the model shows a trend similar to results obtained from the MSFC Bearing and Seal Materials Tester. The technique of relating the two phase heat transfer coefficient to local vapor quality will be applied to the tester model and compared with test data.

The investigation of increased diametrical clearance for the 45 mm bearing was continued. The two phase heat transfer program was used along with a friction factor of 0.25. The clearances studied were 5.9, 6.3, 7.4, and 7.8 mils. These preliminary results showed that the component temperatures and internal operating clearance increased with increased diametrical clearance. These results are not final and the investigation will be continued.

Due to some difficulties with the initial NASTRAN model of the 57 mm inner race a revised NASTRAN model was built. The revised model will simplify the transfer of temperature data from SINDA and also provide smoother stress contour lines. To help simplify the transfer of temperature data even more, a FORTRAN program was written to read the SINDA temperature data and

arrange the temperatures to match the temperature data needed for NASTRAN. The FORTRAN program outputs the temperature data in a format which can be directly added to the NASTRAN model.

## 2.0 SYNOPSIS OF PREVIOUS REPORTS

Previous reports included the items described below.

### 2.1 Second Quarter 1986

- o Transient Thermal Analysis of 57 mm Bearing
- o Transient Thermal Analysis of 45 mm Bearing
- o Sensitivity of 57 mm Bearing to Increased Outer Race Curvature
- o Status of Stress Model of 57 mm Bearing
- o Incorporation of Two-Phase Heat Transfer Coefficient into Bearing Thermal Model
- o Status of ADORE Bearing Analysis Computer Program, and
- o Status of Software Conversion to EADS
- o Sensitivity of 45 mm Bearing to Increased Diametrical Clearance
- o Sensitivity of 45 mm Bearing to Increased Outer Race Curvature
- o Development of Simplified Bearing Thermal/Mechanical Model for the VAX

### 2.2 First Quarter, 1986

- o Status of 45 mm Bearing Thermal Model Investigation,
- o Sensitivity of 57 mm Bearing to Increased Diametrical Clearance,
- o Heat Transfer Coefficients in Two-Phase Flows,
- o Stress Model of 57 mm Bearing Inner Race,
- o Sensitivity of 45 mm Bearing to Increased Diametrical Clearance,
- o Development of Simplified Bearing Thermal/Mechanical Model for the VAX, and
- o Updates to BSMT Data.

### 2.3 1985 Annual Report

- o Development and Use of a Test Condition Data Base,
- o Development and Use of a 45 mm Bearing Mechanical/Thermal Model,
- o LOX Turbopump Pump End 45 mm Bearing Operating Characteristics,
- o SSME LOX Turbopump Bearing Coolant Flow Characteristics,

- o LOX Tester Bearing Friction and Viscous Heat Estimates,
- o Heat Generation for the 60 Hole Diverter - .046" Diameter Holes,
- o Estimate of Ball to Cage Heat Generation,
- o Cage Loads from Coolant Jets and Fluid Friction,
- o Coolant Flow Diverter Velocities,
- o Contact Angles and Ball Tracks as a Function of Bearing Load,
- o LOX Turbopump Turbine End Bearing Thermal Investigation,
- o Sensitivity of 57 mm Bearing to Increased Friction Heat Generation,
- o Ball Excursions as a Function of Loading for the 57mm LOX Turbopump Bearing,
- o Cage Load Stresses for the 57 mm LOX Turbopump Bearing, and
- o Development of Bearing Modeling Program ADORE for LOX Turbopump Bearing.

### 3.0 WORK PERFORMED DURING JULY 1986

This section describes the work performed during the month of July, 1986. Highlights of the July, 1986 efforts are shown in Exhibit 3.1.

### EXHIBIT 3.1 HIGHLIGHTS OF RESULTS OF THE JULY, 1986 REPORTING PERIOD

- o A technique for modifying the two phase heat transfer coefficient has been integrated into and evaluated for both the 45 mm and 57 mm bearing models. Results show that the flow rate affects the bearing operating temperatures more than previously estimated.
- o Increased diametrical clearance was assessed with the modified two phased heat transfer for the 45 mm bearing. For the loading conditions used increased clearance caused an increase in bearing temperatures. This was probably caused by increased contact angles causing increased ball spin and heat generation.
- o Revisions of the NASTRAN model of the 57 mm inner race to simplify the transfer of temperature data from the SINDA model have been made.



### 3.1 Evaluation of 45 mm Bearing Model Two Phase Heat Transfer Program

The modified heat transfer program has been integrated into the 45 mm bearing model code. The heat transfer program predicts a ratio of experimental to calculated heat transfer coefficients which includes vapor quality effects. The experimental heat transfer to calculated ratio is determined using the Hendricks correlation which is a function of quality. This ratio is multiplied by the heat transfer coefficient previously calculated for the surface nodes of the bearing assuming vapor properties evaluated at the film temperature.

Currently, four cases were used to evaluate the effects of this method on bearing temperatures. The cases used flow rates of 3.6 and 7.0 lbs/sec and inlet temperatures of -230°F and saturated. A pressure drop of 40 psia was used for the 7 lbs/sec case and 30 psia for 3.6 lbs/sec. An outlet pressure of 304 psia and a friction factor of 0.25 was used for both flow rates.

Exhibit 3.1.1 shows the results of the evaluation. The flow rate and subcooling have a greater affect on the steady-state operating temperatures than was previously modeled. The increase in flow rate from 3.6 to 7.0 lbs/sec decreased the average ball temperature 102°F. Using the subcooled liquid of -230 at 7.0 lbs/sec decreased the average ball temperature by 63°F when compared to the saturated coolant. The computer model was unable to determine a solution for the saturated case using 3.6 lbs/sec. The minimum flow rate which will enable this case to converge is currently under investigation. These modeling techniques should be considered preliminary at this time. Efforts are underway to apply this technique to the bearing tester model and compare results with test data.

### 3.2 Sensitivity of 45 mm Bearing to Increased Diametrical Clearance

Increased diametrical clearance was investigated for the 45 mm bearing with the two phase heat transfer program. The clearances studied were 5.9, 6.3, 7.4, and 7.8 miles with a 0.25 friction factor, 7.0 lbs/sec flow rate, 480 lbs pre load, and saturated coolant.

The average component and maximum track temperatures are listed in Exhibit 3.2.1. The temperatures increase with increasing clearance. Exhibit

# Exhibit 3.1.1 45mm Steady-State Operating Temperatures

COOLANT FLOWRATE	COOLANT TEMP. AT ENTRANCE TO BEARING 1									
	-230 °F					SATURATED*				
	AVERAGE TEMPERATURE (°F)		MAXIMUM TRACK TEMPERATURE (°F)		AVERAGE TEMPERATURE (°F)		MAXIMUM TRACK TEMPERATURE (°F)			
	INNER RACE	BALL	OUTER RACE	INNER RACE	BALL	OUTER RACE	INNER RACE	BALL	OUTER RACE	INNER RACE
3.6	-70	144	-62	298	325	287	X	X	X	X
7.0	-120	42	-70	174	200	219	-81	105	-72	258
										256

\*Saturated Temp. at 7.0 lbs/sec ( $\Delta P=40$  psi) = -215 °F

\*Saturated Temp. at 3.6 lbs/sec ( $\Delta P=30$  psi) = -216 °F

X Thermally Unstable

o Pressure at exit of Bearing 2 = 304 psia

o Friction Factor = 0.25

o 480 lb Axial Preload

Exhibit 3.2.1 Effect of Diametrical Clearance on Component Temperatures

DIAMETRICAL CLEARANCE (mils)	AVERAGE COMPONENT TEMPS. (°F)			MAXIMUM TRACK TEMPS. (°F)		
	INNER RACE	BALL	OUTER RACE	INNER RACE	BALL	OUTER RACE
5.9	-84	100	-73	249	269	252
6.3	-81	105	-72	257	276	256
7.4	-50	168	-53	348	360	309
7.8	-38	192	-46	382	391	326

7.0 lbs/sec FLOW RATE  
 SATURATED COOLANT  
 304 psia AT BEARING 2  
 0.25 FRICTION FACTOR  
 480 lb AXIAL PRELOAD

3.2.2 shows the inner and outer contact stress and the bearing internal operating clearance. The stresses changed only slightly and the internal operating clearance increased with diametrical clearance. This investigation is continuing and these results should be considered preliminary.

### 3.3 Evaluation of 57 mm Bearing Model Two Phase Heat Transfer Program

Integration of the modified heat transfer program into the 57 mm bearing model has been completed. The program has been evaluated using four cases. The cases used flow rates of 4.6 and 9.0 lbs/sec and -245°F and saturated inlet coolant temperatures. Pressure drops of 40 and 30 psia were used for the 9.0 and 4.6 lbs/sec flow rates, respectively. An outlet pressure of 304 psia was used.

The results are listed in Exhibit 3.3.1. They show that the steady-state operating temperatures are greatly affected by coolant flow rate and inlet temperature. By increasing the coolant flow rate from 4.6 to 9.0 lbs/sec the average ball temperature was decreased by 60°F and 46°F for the subcooled and saturated cases, respectively. The average ball temperature was decreased 57°F and 43°F for flow rates 9.0 and 4.6 lbs/sec, by decreasing the inlet coolant temperature from saturation to -245°F. The model now shows a greater effect of coolant flow rate which more closely compares to the experimental data from the bearing tester. As with the 45 mm model, these results should be considered preliminary. The temperatures quoted should at this time be considered trend data. Additional analysis is needed to investigate additional effects such as increased friction and cage heating.

### 3.4 Status of the Stress Model of the 57 mm Bearing

The NASTRAN stress model of the 57 mm bearing was constructed to match the geometry of the SINDA thermal model as closely as possible. The stress contour lines on the NASTRAN model were not as smooth as should be due to the means for determining the grid point coordinates of the model. Because of the difficulty of transferring temperature data to the NASTRAN model and the type of contour lines produced by the model, modification were made to update the model.

Exhibit 3.2.2 Effect of Diametrical Clearance on Bearing Stresses and Operating Clearances

DIAMETRICAL CLEARANCE (mils)	CONTACT STRESS (psi)		INTERNAL OPERATING CLEARANCE (mils)
	INNER	OUTER	
5.9	$4.86 \times 10^{-5}$	$3.62 \times 10^{-5}$	3.82
6.3	$4.86 \times 10^{-5}$	$3.62 \times 10^{-5}$	4.22
7.4	$4.93 \times 10^{-5}$	$3.67 \times 10^{-5}$	4.72
7.8	$4.94 \times 10^{-5}$	$3.69 \times 10^{-5}$	4.91

7.0 lbs/sec FLOW RATE  
 SATURATED COOLANT  
 304 psia AT BEARING 2  
 0.25 FRICTION FACTOR  
 480 lb AXIAL PRELOAD

### Exhibit 3.3.1 57mm Steady-State Operating Temperatures

COOLANT FLOWRATE		COOLANT TEMP. AT ENTRANCE TO BEARING 4											
		-245°F						SATURATED*					
		AVERAGE TEMPERATURE (°F)			MAXIMUM TRACK TEMPERATURE (°F)			AVERAGE TEMPERATURE (°F)			MAXIMUM TRACK TEMPERATURE (°F)		
lbs/sec	INNER RACE	BALL	OUTER RACE	INNER RACE	BALL	OUTER RACE	INNER RACE	BALL	OUTER RACE	INNER RACE	BALL	OUTER RACE	
4.6	-93	198	-68	378	525	185	-75	241	-43	405	567	218	
9.0	-119	138	-101	338	464	140	-88	195	-64	375	520	186	

\* Saturation Temp. for 4.6 lbs/sec = -216°F

\* Saturation Temp. for 9.0 lbs/sec = -215°F

◦ 3000 lb Axial Load

◦ 0.2 Friction Factor

During this reporting period an updated NASTRAN model was built. The grid point coordinates were calculated based on the same geometry used to build the SINDA model. The grid points in the NASTRAN model were located at the same points as the node centers in the SINDA model.

Along with changing the location of the grid points, additional detail was added to the model. Additional detail was added because the stress contour lines are sensitive to the size and type of structural elements used. The main area of interest at this time is directly under the contact area and where high temperature gradients are present. Exhibit 3.4.1 shows a comparison of the initial and updated NASTRAN finite element models. Exhibit 3.4.1 also shows the added detail to the updated NASTRAN model.

The location of grid points permits a direct transfer of temperature data. The temperature of grid points which do not correspond to a node location on the SINDA model were taken to be the average of two grid points which formed a line intersecting the unknown grid point temperature. Grid point temperatures which could not be determined by averaging were assumed to be equal to the node temperature which correspond to the grid point location.

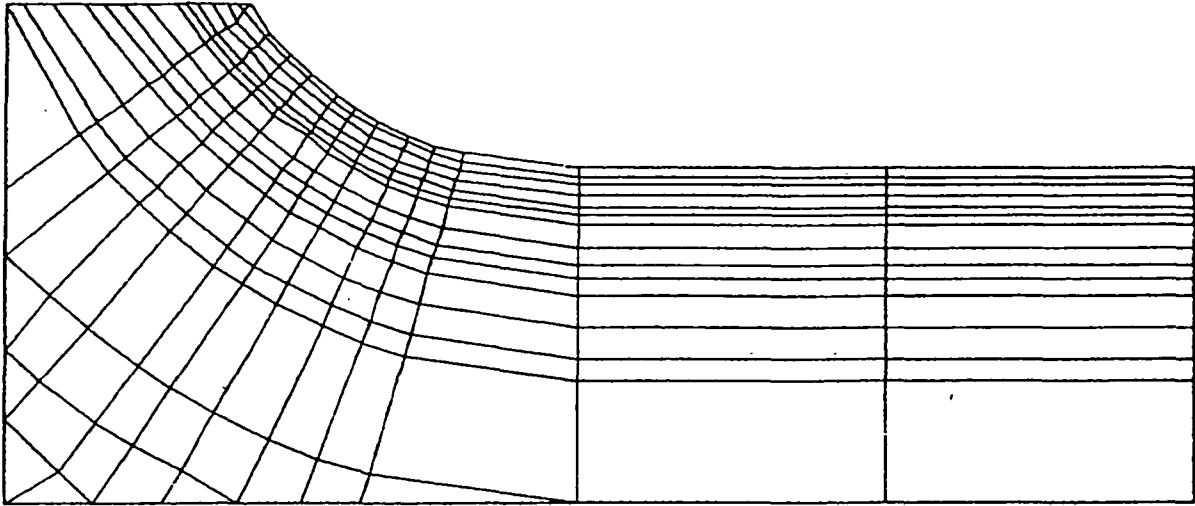
To obtain the temperature data the output of SINDA was slightly modified to write the temperature data needed to a special file. A simple FORTRAN program was written to read the SINDA temperatures from a file and arrange the temperature data to match the temperature data needed for NASTRAN. The FORTRAN program data also outputs the NASTRAN temperatures in the correct format to be directly added to the NASTRAN model.

The initial run of the updated NASTRAN model with pressure and thermal loads indicated a slight problem. This problem should be solved and the NASTRAN model debugged during the next reporting period.

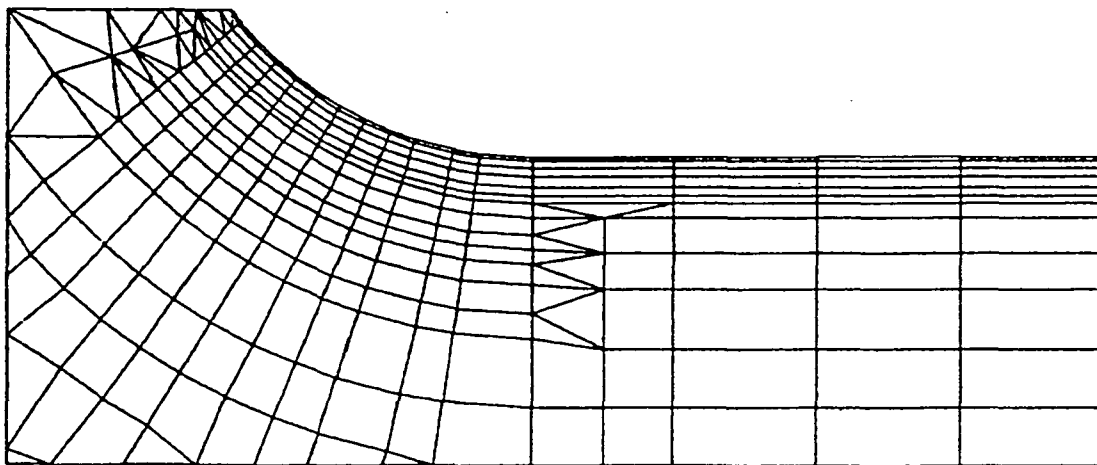
### 3.5 Modified Practice of Evaluating Two Phase Heat Transfer Coefficients

The evaluation of heat transfer coefficients for two phase flow is continuing. The heat transfer from metal surface to bulk flow of fluid is a function of the surface to fluid heat transfer coefficient. For the inner and outer race, the heat transfer coefficient is determined from the Dittus-Boelter equation

$$\frac{hD}{k} = 0.023 Re^{0.8} Pr^{0.4} \quad (1)$$



a) Initial NASTRAN model



b) Revised NASTRAN model

Exhibit 3.4.1 Comparison between the initial and revised NASTRAN finite element models



For the ball, Katsnellson's equation is used

$$\frac{hD}{k} = 2 + 0.03 \text{ Pr}^{0.33} \text{ Re}^{0.54} + 0.35 \text{ Pr}^{0.356} \text{ Re}^{0.30} \quad (2)$$

In these equations, fluid properties are evaluated at the average film temperature. Liquid properties are used when the wall temperature is less than 5°F super heat. Otherwise vapor properties are used to evaluate the local heat transfer coefficient from appropriate equations.

In the modified practice, a scaling factor is calculated as a function of quality. The scaling factor is determined from the following correlation (Reference 1).

$$\frac{\text{Nu}_{\text{exp}}}{\text{Nu}_{\text{calc},v,s.p}} = \exp \left[ -0.185 - 0.251 \ln x - 0.00767 (\ln x)^2 \right] \quad (3)$$

where  $x$  is the mass quality of vapor and

$$\text{Nu}_{\text{exp}} = \frac{hD}{k} \quad (4)$$

$$\text{Nu}_{\text{calc},v,s.p} = 0.026 \text{ Re}_{v.s.p.}^{0.80} \text{ Pr}_v^{0.33} \left( \frac{\mu_v}{\mu_w} \right)^{0.14} \quad (5)$$

$$\text{Re}_{v.s.p.} = \frac{\rho_v U_{\text{avg}} D}{\mu_v} \quad (6)$$

$$\text{Pr}_v = \frac{C_{p,v} \mu_v}{K_v} \quad (7)$$

From equation (5)  $h_{\text{calc}}$  is calculated and the scaling factor,  $s_f$ , is defined by

$$s_f = \frac{h}{h_{\text{calc}}} \quad (8)$$

In the SINDA thermal model, the heat transfer coefficients evaluated from equations (1) and (2) are updated by the scaling factor

$$h_{T.p} = h s_f \quad (9)$$

Exhibit 3.5.1 shows the variation of  $h_{T.p}$  with quality at three different flow rates (i.e., 7.2, 3.6 and 1.8 lb/s respectively). The two phase heat transfer coefficients are plotted at constant surface temperatures. At lower temperatures, the heat transfer coefficient is a stronger function of flow rate than at higher temperatures. At very high temperatures,  $h_{Tp}$  remains fairly constant with the flow rate. The flow rate has little influence on the heat transfer coefficient when complete vapor blanketing of the surface occurs.

#### 4.0 ANTICIPATED WORK

The two phase heat transfer program will continue to be used with both the 45 and 57 mm models to determine the bearings operating condition. Operating conditions of the MSFC bearing tester will be modeled using the modified heat transfer methods to compare results with test data. The study of bearing sensitivity to increased diametrical clearance will be continued.

A program which will provide the capability to input bearing preload will be incorporated into the bearing model. This program will also be able to model the movement of the bearings due to elastic components such as springs.

During the next reporting period, work will be done to debug the revised NASTRAN model. After the revised model is running, a more detailed analysis of the results will be done.

#### 5.0 REFERENCES

Reference 1 E.G. Brentari, P.J. Giarrantano and R.V. Smith , "Boiling Heat Transfer for Oxygen, Nitrogen, Hydrogen, and Helium"  
NBS Technical Note NO. 317, September 1965.

# TWO PHASE HEAT TRANSFER COEFFICIENTS IN BEARING

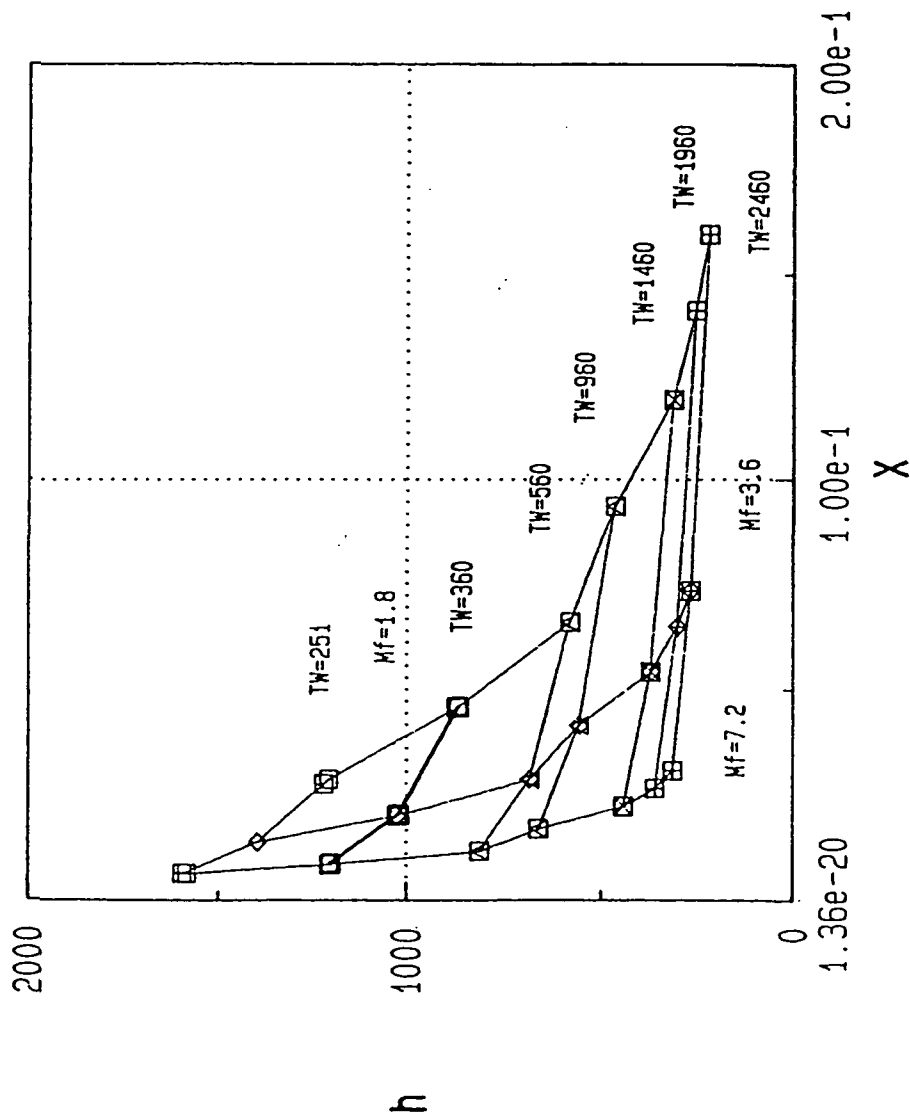


Exhibit 3.5.1